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A thermodynamic analysis of waste heat recovery from reciprocating engine power plants by means of Organic Rankine Cycles

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HIGHLIGHTS

- Waste heat recovery potential of reciprocating engines was studied.
- Thermodynamic optimization for ORCs was carried out with different fluids.
- The utilization of exhaust gas and charge air heat is presented and discussed.
- Simplified economic feasibility study was included in the analysis.
- Power increase of 11.4% was obtained from exhaust gas and 2.4% from charge air.

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ABSTRACT

Organic Rankine Cycle (ORC) is a Rankine cycle using organic fluid as the working fluid instead of water and steam. The ORC process is a feasible choice in waste heat recovery applications producing electricity from relatively low-temperature waste heat sources or in applications having a rather low power output. Utilizing waste heat from a large high-efficiency reciprocating engine power plant with ORC processes is studied by means of computations. In addition to exhaust gas heat recovery, this study represents and discusses an idea of directly replacing the charge air cooler (CAC) of a large turbocharged engine with an ORC evaporator to utilize the charge air heat in additional power production. A thermodynamic analysis for ORCs was carried out with working fluids toluene, *n*-pentane, R245fa and cyclohexane. The effect of different ORC process parameters on the process performance are presented and analyzed in order to investigate the heat recovery potential from the exhaust gas and charge air. A simplified feasibility consideration is included by comparing the ratio of the theoretical heat transfer areas needed and the obtained power output from ORC processes. The greatest potential is related to the exhaust gas heat recovery, but in addition also the lower temperature waste heat streams could be utilized to boost the electrical power of the engine power plant. A case study for a large-scale gas-fired engine was carried out showing that the maximum power increase of 11.4% was obtained from the exhaust gas and 2.4% from the charge air heat.

1. Introduction

The environmental impacts of energy production and high fuel prices highlight the demand for developing more efficient and environmentally friendly energy systems. In reciprocating engine power plants, a large proportion of fuel power is not converted into electrical power but is removed from the engine in the form of waste heat. The efficiency of the engine system could be improved by converting these waste heat streams into additional electricity. The conventional steam cycle has been widely used to utilize high-temperature exhaust gas waste heat in different heat recovery and combined cycle applications. However, a steam Rankine cycle does not allow efficient recovery of low-temperature waste heat at temperature levels below ~370 °C [1]. One of the most promising technologies in low-temperature heat utilization is related to Organic Rankine Cycles (ORC). ORC is a Rankine cycle which uses organic fluid as the working fluid instead of water and steam. The main components of a simple ORC are the turbine, evaporator, generator, condenser, and pump. ORC processes are suitable in producing electricity from relatively low-temperature waste heat

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Nomenclature

Latin alphabet

A	heat transfer area, m^2
P	power output, kW
p	pressure, bar
q_m	mass flow rate, kg/s
T	temperature, °C, K
U	overall heat transfer coefficient, W/m^2 K

Greek alphabet

ε	recuperator effectiveness
η	efficiency
ϕ	heat rate, kW

Subscripts

c	condensing
co	condenser
cr	critical
e	electrical
ev	evaporator, evaporation
fp	feed pump
gm	generator and mechanical
in	inlet
l	liquid
LMTD	Log Mean Temperature Difference
max	maximum
mis	miscellaneous
net	net
out	outlet
pp	pinch-point
re	recuperator
sh	superheating
t	turbine
theo	theoretical
tot	total
v	vapor

Abbreviations

CA	Charge Air
CAC	Charge Air Cooler
EG	Exhaust Gas
ORC	Organic Rankine Cycle

sources and in applications having a rather small power output [1,2,3]. Typical existing ORC applications are geothermal energy, biomass applications, solar energy and the recovery of waste heat [4,5].

The ORC working fluids can be generally divided into three main categories: 1) dry, 2) isentropic, and 3) wet [1,6]. For most of the organic fluids, the expansion process in the turbine occurs in the dry region, and therefore, there is no risk of liquid drops damaging the turbine [6]. Saleh et al. [7] carried out a study using 31 different working fluids for low-temperature ORC processes. They concluded that the superheating of the working fluid does not improve the thermal efficiency of the ORC cycle with fluids having overhanging saturated vapor line in the temperature versus entropy *T,s*-diagram. Thus, the need for superheating should be evaluated based on the working fluid properties and the evaporator type used in the process. In most of the cases, a single-stage turbine can be used because of the low enthalpy drop over the turbine with organic fluids [2]. When using dry fluids, the working fluid temperature after the turbine is relatively high and the working fluid exits the turbine at superheated state. Therefore, a recuperator can be used for preheating the liquid working fluid entering the evaporator with low-pressure vapor exiting the turbine, in order to increase the obtained efficiency of the ORC process [3]. An example of a typical ORC process with the recuperator is presented in Fig. 1 and the principle of a subcritical, superheated and recuperated ORC process is presented on *T,s*-diagram in Fig. 2. According to Figs. 1 and 2, the ORC process can be divided into the following stages: expansion in the turbine (1–2), desuperheating in the recuperator and in the condenser (2–4), condensing in the condenser (4–5), pressure rise in the feed pump (5–6), preheating in the recuperator and in the evaporator (6–8), evaporation (8,9) and superheating (9–1) in the evaporator.

The selection of the working fluid and the selection of the ORC working parameters, as well as the cycle configuration, can be considered as the most important steps in designing ORC systems. Quoilin et al. [8] studied the optimization of a small-scale ORC for waste heat recovery applications, taking into account both the thermodynamic and economic aspects. They concluded that the thermodynamic optimization can be used for evaluating the suitable fluid candidates, but does not necessarily result in the selection of a fluid, that would be the optimal from the economic point of view. Branchini et al. [9] considered six ORC performance

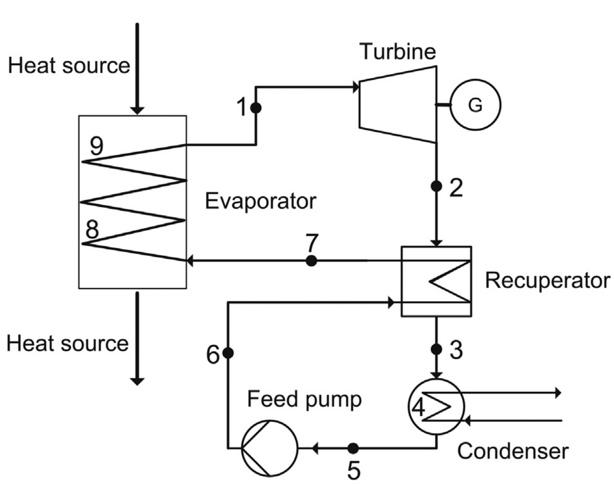

Fig. 1. Example of a typical ORC process with a recuperator.

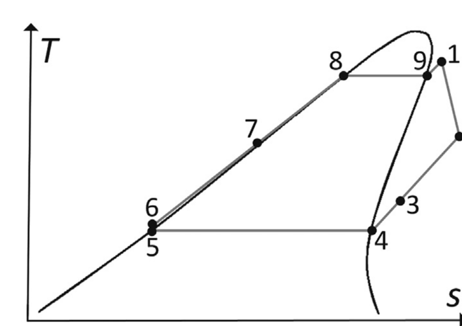

Fig. 2. The principle of a superheated ORC process on *T,s*-diagram.

indexes, namely the ORC cycle efficiency, ORC specific work, volumetric expansion ratio over the turbine, ORC working fluid to heat source fluid mass flow ratio, recovery efficiency, as well as the heat exchanger surfaces of the ORC. They proposed these indexes being beneficial in the selection of working fluid, as well as in determining suitable process parameters for ORCs, in terms of both performance and economics. Hettiarachchi et al. [10] presented optimum design criteria for a low-temperature ORC, based on the ratio of the total heat transfer area and the net power output, to evaluate the costs of the ORC process.

Angelino and Moroni [11] made one of the first studies on recovering waste heat of different prime movers. They concluded that by using organic fluid instead of steam, a larger amount of heat can be exctracted to the waste heat process, when a relatively low-tempeature exhaust gas stream was used as a heat source. Angelino and Moroni [11] reported that the combined cycle efficiencies around 47% are achievable from low-speed diesel engines, and by using the ORC as a bottoming cycle it was possible to generate 12% additional power. More recent studies on recovering waste heat from large-scale reciprocating engines have been carried out, for example by Bombarda et al. [12] as well as by Vaja and Gambarotta [13], concentrating mainly on the utilization of the exhaust gas heat. Bombarda et al. [12] compared ORC and Kalina processes utilizing exhaust gas heat streams from two 8.9 MWe diesel engines. Their results showed that ORC technology has advantages in this kind of applications due to lower pressure levels, simpler turbine, and less corrosive fluids when compared to Kalina cycles. The thermodynamic efficiencies of the Kalina and ORC processes obtained from the simulations were 19.7% and 21.5%, respectively. The electrical power produced using the Kalina cycle was 1615 kW, and using the ORC, a power output of 1603 kW was achieved. The electrical efficiency of the studied diesel engine was 46.0% without a bottoming cycle, and the electrical efficiencies of 50.2% and 50.1% were obtained using the Kalina cycle and the ORC as the bottoming cycle, respectively. The working fluid used in the ORC was hexamethyldisiloxane (MM), and a recuperator was used. According to Bombarda et al. [12] also the heat streams from jacket cooling water, charge air and lube oil system are available, but no further analysis utilizing these low-temperature heat sources was conducted. The waste heat recovery from a supercharged natural gas-fired medium-speed engine was studied by Vaja and Gambarotta [13]. The electrical power of the engine was approximately 3 MW, the exhaust gas temperature around 470 °C and the rated electrical efficiency of the engine was 41.8%. It was evaluated that there was a capacity of 1700 kW available by cooling the exhaust gases to 120 °C and approximately 1000 kW from the engine cooling water. Three different ORC cycle configurations were adopted to utilize the waste heat streams. First, using a simple ORC cycle utilizing exhaust gases only; second, using a simple ORC cycle with the engine cooling water implemented to preheat the organic fluid; and third, using an ORC cycle with a recuperator utilizing exhaust gases only. Also three different working media were studied; namely benzene, R11 and R134a. Vaja and Gambarotta [13] concluded that it was possible to achieve a significant efficiency increase using the ORC as a bottoming cycle. The highest electrical efficiency, 47.1%, was achieved using benzene as a working fluid, and using the ORC cycle with a recuperator or the ORC cycle with preheating. Vaja and Gambarotta [13] also concluded that using preheating leads to a simpler heat exchanger compared to the use of a recuperator.

Exhaust gas heat recovery in smaller power output systems by means of ORCs were studied, for example by Invernizzi et al. [14] and Uusitalo et al. [15]. The exhaust gas heat utilization from a micro gas turbine by means of ORC was studied by Invernizzi et al. [14]. The electric power output of the micro gas turbine was 100 kW and the exhaust gas temperature level ranged between 250 and 300 °C. Many different working fluids were considered, including e.g. siloxanes and hydrocarbons. They noticed that the thermodynamic efficiency of the ORC cycle increased as the molecular complexity of the working fluid increased, but the more complex molecular complexity led to lower recovery efficiency, i.e. higher exhaust gas outlet temperatures. Therefore, it was concluded that it would be beneficial to use a rather low molecular complexity working fluid. Invernizzi et al. [14] also discussed the preliminary design of ORC process components. The designed turbine was a 40 kW two-stage axial turbine and hexamethyldisiloxane (MM) was used as a working fluid. It was also estimated that adding the ORC as a bottoming cycle to the micro gas turbine would increase the specific cost of the system from 1.2 k€/kW to 1.6–1.8 k€/kW when a specific cost of the ORC is 2.5–3 k€/kW. Uusitalo et al. [15] analyzed the exhaust gas heat recovery from a 120 kW diesel engine by adopting mainly siloxanes as ORC working fluids. They concluded that the use of fluids having a high molecular weight, leads to high pressure and volumetric ratios over the turbine, which enables the use of larger turbine wheels and lower rotational speeds, compared to fluids having a lower molecular complexity. They concluded that the use of working fluids having a relatively high molecular weight can be beneficial, especially in realizing turbomachinery for small power capacity ORC systems.

In large reciprocating engines, the most significant waste heat stream can be identified as exhaust gas heat. There are also lower temperature waste heat streams, mainly heat in the pressurized charge air after the turbocharger, as well as heat in engine jacket cooling water and lubrication oil cooling circuit. The aim of this computational study is to investigate the potential of waste heat recovery from exhaust gases (EG) and charge air (CA) by means of ORC. Despite previous studies related to the utilization of waste heat of large-scale reciprocating engines, studies analyzing the possibility of utilizing the charge air heat are lacking at the moment. This study represents and discusses the possibility of replacing the charge air cooler (CAC) of a large turbocharged engine with an ORC evaporator, to utilize the heat from c. 180–220 °C charge air in additional power production, as well as to study the overall power production improvement potential by means of heat recovery. The studied engine combined cycle is presented in Fig. 3. The utilization of the heat from engine jacket water cooling and lubrication oil cooling were excluded in this study, due to low temperature level of about 90 °C. The ORC simulations were performed with several heat source temperatures and the effect of minimum pinch-point temperature difference in the evaporator, as well as the effect of condensing temperature on the process performance was included into the analysis. A simplified feasibility evaluation was included by comparing the ratio of the theoretical heat transfer areas needed and the obtained power output of ORC processes, as suggested in [10]. Finally, a case study of waste heat utilization for a 16.6 MW gas-fired engine was carried out.

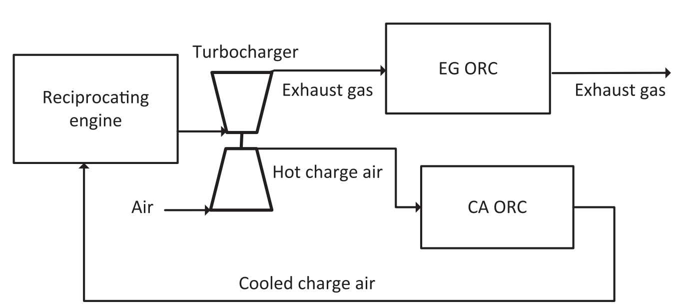

Fig. 3. The principle of the studied engine combined cycle. Exhaust gases (EG) and charge air (CA) heat are utilized by means of ORC.

2. Methods and process analysis

The thermodynamic analysis of the waste heat recovery potential from large reciprocating engines was carried out by performing ORC system optimization for the selected waste heat streams of reciprocating engines. The studied waste heat stream temperatures, typical to large-scale turbocharged engines, were 300–400 °C for the exhaust gas and 180–220 °C for the charge air.

The ORC performances were studied with varying heat source inlet and outlet temperatures. The ORC analysis method is based on the thermodynamic process calculation principles of the Rankine process. The thermodynamic state of the working fluid was determined at the inlet and outlet of each process component, as well as the saturated vapor and liquid states in the condenser and in the evaporator. The heat source values, process condensing temperature and component efficiencies were given as an input. The cycle power output, efficiency, thermodynamic states of the working fluid and mass flow rate were obtained as a result of the simulation by solving the continuity and energy equation for each process component. Commercial software FluidProp 2.3 [16] using thermodynamic and transport properties database Refprop [17] was used for determining the working fluid thermodynamic properties. Toluene, *n*-pentane, R245fa and cyclohexane were selected as the working fluids for the analysis and the Helmholz equation of state was used to solve the thermodynamic properties of each working fluid [18,19,20]. The working fluid selection was based on suitable thermodynamic characteristics of the fluids including dry expansion in the turbine, relatively high cycle performance and the critical temperature close to the waste heat source temperature. The main thermodynamic properties of the selected working fluids are shown in Table 1.

The analysis was performed for slightly superheated process and the evaporation pressure was limited to be subcritical $p_{ev}/p_{cr} = 0.95$ and thus, no supercritical fluid conditions were considered. The minimum pinch-point temperature difference in the ORC evaporator was limited to 15 °C and condensing temperature of 50 °C was used. The effect of the minimum pinch-point temperature difference in the evaporator and the effect of different condensing temperatures on the ORC performance are also presented and discussed in the study. The evaporation pressure of the process was optimized based on the criteria that the minimum allowed pinch-point temperature difference in the evaporator was reached, or in case the temperature difference in the evaporator remained sufficient, the limit of $p_{ev}/p_{cr} = 0.95$ was used for the evaporation pressure. The main process specifications and component efficiencies used in the process analysis are shown in Table 2.

The recuperator effectiveness, presented in Table 2, is defined as

$$\varepsilon = \frac{T_{v,in} - T_{v,out}}{T_{v,in} - T_{l,in}} \tag{1}$$

The effects of pressure losses were excluded from the ORC analysis and thus, the results show slightly higher performance for the cycles than in a real process. Especially, the pressure losses occurring between the turbine outlet and the condenser have an effect on the cycle performance. The pressure losses are dependent on heat exchanger and pipeline geometries, as well as on working fluid density and flow velocity at the turbine outlet. It was also assumed that a liquid coolant is available for condenser cooling without a need for cooling air fans. If the air fans are needed for condenser cooling, the auxiliary power consumption would be higher than used in the analysis, leading to slightly lower net power outputs than obtained in this study.

The ORC electric power outputs obtained from the simulations were compared to the theoretical heat transfer areas needed in the processes in order to make a simplified estimation of the feasibility of the process. The theoretical heat transfer areas required in different processes were evaluated based on the temperature profiles of the heat exchangers. The evaluation of theoretical heat transfer areas were based on the log mean temperature difference in the heat exchangers and overall heat transfer coefficients. The theoretical heat transfer area is defined as

$$A = \frac{\phi}{U\Delta T_{LMTD}} \tag{2}$$

The total theoretical heat transfer area is defined as

$$A_{tot} = A_{ev} + A_{co} + A_{re}. \tag{3}$$

It should be noted that the aim is to compare the effect of adopting different working fluids and different process working conditions and parameters on theoretical heat transfer areas, rather than accurately estimate the heat transfer areas needed in an actual process. Thus, the selection of the heat exchanger type and the flow arrangement in the heat exchanger was not taken into account in the analysis. The assumption of constant heat transfer coefficients was used in the analysis for all the selected fluids. The overall heat transfer coefficients used in the analysis are shown in Table 3 and they were evaluated based on reference [21].

3. Results and discussions

3.1. Exhaust gas heat utilization

The exhaust gas heat utilization analysis was performed using exhaust gas temperatures of 300, 350 and 400 °C, typical for large-

Table 1
Working fluid properties [18,19,20].

Fluid	Chemical formula	Molecular weight [kg/kmol]	Critical temperature [°C]	Critical pressure [bar]
Toluene	C_7H_8	92.1	318.6	41.3
n-pentane	C_5H_{12}	72.1	196.6	33.7
R245fa	$C_3H_3F_5$	134.1	154.1	36.5
Cyclohexane	C_6H_{12}	84.2	280.5	40.8

Table 2
Component efficiencies and process specifications used in the process calculations.

η_t	80	%
η_{gm}	90	%
η_{fp}	60	%
$\eta_{fp,m}$	85	%
$p_{ev,max}/p_{cr}$	95	%
T_{sh}	10	K
ε	0.6	–
$P_{e,mis}/P_e$	2	%

Table 3
Overall heat transfer coefficients.

Heat exchanger	U [W/m^2 K]
Evaporator:	
Organic fluid – heat source	20
Recuperator:	
Organic vapor – organic liquid	40
Condenser:	
Organic vapor – liquid cooling fluid (desuperheating)	60
Condensing organic vapor – liquid cooling fluid	900

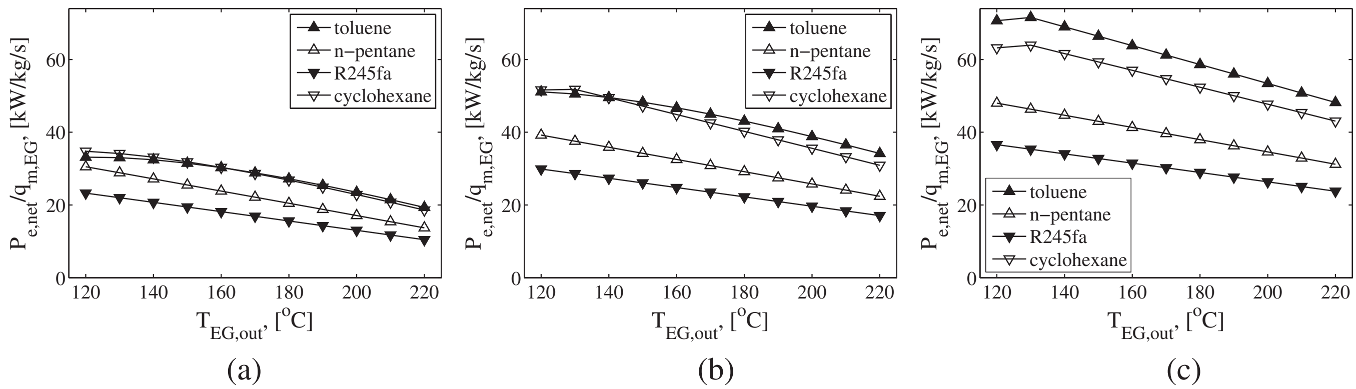

Fig. 4. The effect of exhaust gas temperature at the outlet of the evaporator on the ratio of net power output to exhaust gas mass flow rate for different organic working fluids with exhaust gas inlet temperatures of (a) 300 °C, (b) 350 °C, (c) 400 °C.

scale turbocharged engines. The results of the ORC process power outputs are presented in Fig. 4a, b and c. Toluene and cyclohexane represents the highest performance from the four selected fluids in most of the cases. The highest power output $P_{e,net}/q_{m,EG} = 71.6$ kW/kg/s was achieved with toluene for EG inlet temperature of 400 °C and $P_{e,net}/q_{m,EG} = 51.8$ kW/kg/s and 34.8 kW/kg/s with cyclohexane for EG inlet temperatures of 350 °C and 300 °C. Fluids with higher critical temperature, toluene and cyclohexane, represent a significantly higher performance than lower critical temperature fluids, *n*-pentane and R245fa. As can be observed form the results, the obtained power output increases as the exhaust gas outlet temperature decreases or exhaust gas inlet temperature increases. This can be explained by a larger amount of heat utilized in the evaporator. The power outputs for the selected fluids represent a linear behavior with different exhaust gas outlet temperatures if a sufficient temperature difference is maintained in the evaporator with the highest allowed pressure $p_{ev}/p_{cr} = 0.95$ of the fluid. With the exhaust gas inlet temperature of 300 °C or low exhaust gas outlet temperatures, the evaporation pressure have to be lowered, with fluids toluene and cyclohexane, to maintain the sufficient temperature difference in the evaporator. This can be observed as a change in the slope of the curve in Fig. 4(a), (b) and (c). The fluids with a high critical point, toluene and cyclohexane, can be better matched to the exhaust gas temperature than the fluids R245fa and *n*-pentane, which are restricted by the lower critical point. With

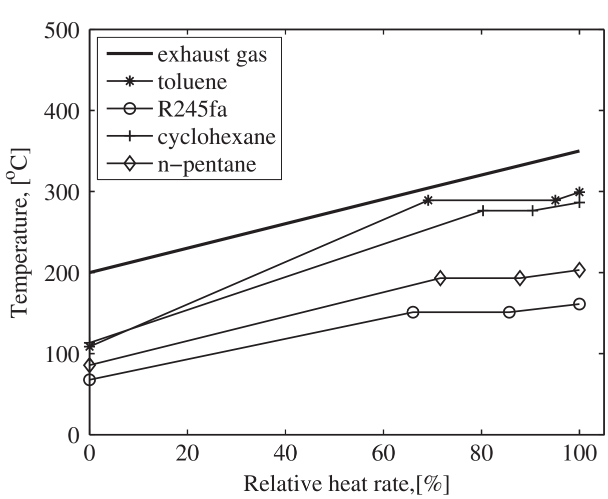

Fig. 5. An example of temperature profiles in the evaporator in 350 °C exhaust gas utilization.

reference to this, an example of temperature profiles of the selected working fluids in the evaporator are shown in Fig. 5.

The results for the ratios of theoretical heat transfer area compared to the net electric power output are shown in Fig. 6(a), (b) and (c). The higher the exhaust gas inlet temperature or the exhaust gas outlet temperature, the more economically feasible processes are obtained based on this ratio, due to larger temperature differences between the working fluid and exhaust gas. The lowest exhaust gas outlet temperatures lead to the highest power output of the cycle but rather large heat transfer areas are needed, especially with fluids cyclohexane and toluene. This can be explained by smaller temperature differences in the evaporator and larger heat rate in the evaporator, if the maximum power output is desired. The working fluids with lower critical point, namely R245fa and *n*-pentane, introduce smaller ratios with low EG inlet temperatures and low EG outlet temperatures, when compared to toluene and cyclohexane.

It should be noted that if a fuel with relatively high sulfur content is used in the reciprocating engine, the exhaust outlet temperature should be limited above the acid dew point of the exhaust gases, in order to prevent the corrosive effects occurring in the ORC evaporator and in the exhaust smoke stack.

3.2. Charge air heat utilization

In modern industrial diesel and gas engines, high pressure ratios are typically adopted for the turbocharger in order to improve the engine efficiency and increase the power output. The turbocharger pressure ratio highly affects the charge air outlet temperature after the turbocharger compressor and thus, the turbocharger pressure ratio has a major impact on waste heat recovery potential from the charge air. The charge air temperature range used in this study is 180–220 °C, which is corresponding to the engine turbocharger pressure ratios in a range of 3.5–4.5. The results for ORC net power outputs are presented in Fig. 7(a), (b) and (c). The highest power outputs of $P_{e,net}/q_{m,CA} = 16.2$ kW/kg/s, 11.6 kW/kg/s and 7.7 kW/kg/s with CA inlet temperatures of 220 °C, 200 °C and 180 °C were obtained with working fluid R245fa. Based on the results R245fa and *n*-pentane introduce the highest performance with low CA outlet temperatures and toluene and cyclohexane with high CA outlet temperatures. The peak value for the power output can be found for different working fluids with different charge air outlet temperatures. The working fluid R245fa reaches the pressure limit of $p_{ev}/p_{cr} = 0.95$ with high charge air outlet temperatures. With low charge air outlet temperatures the evaporator pressure level is lower to maintain a sufficient pinch-point temperature difference

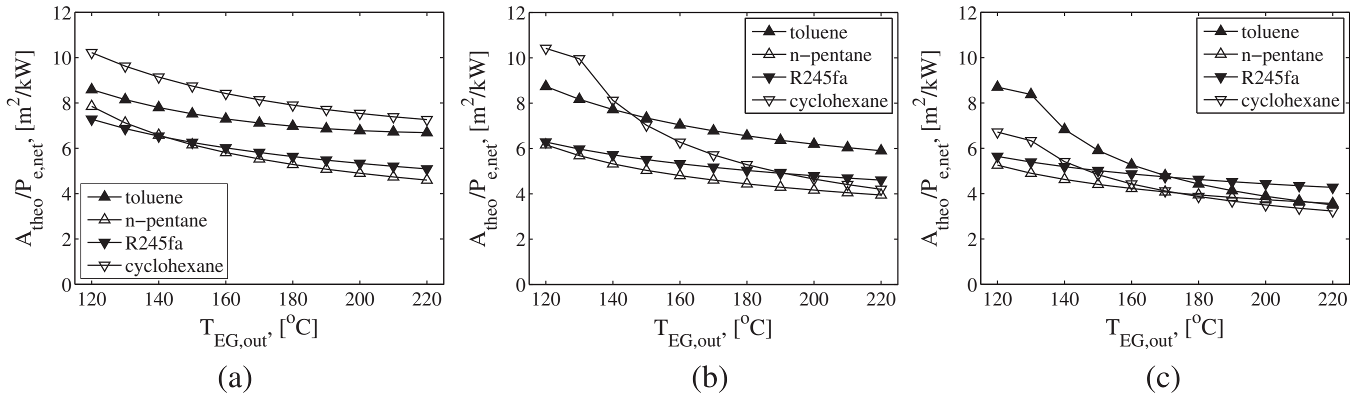

Fig. 6. The effect of exhaust gas temperature at the outlet of the evaporator on the ratio of theoretical heat transfer area to net power output for different organic working fluids with exhaust gas inlet temperatures of (a) 300 °C, (b) 350 °C, (c) 400 °C.

in the evaporator. This can be observed as a change in the slope of the curve with R245fa in Fig. 7(a), (b) and (c).

An example of the evaporator temperature profiles in 200 °C charge air utilization is shown in Fig. 8. With working fluids toluene and cyclohexane a larger portion of the heat is needed for the fluid evaporation than with fluids R245fa and *n*-pentane. This can be explained by the lower evaporation temperature and pressure levels compared to the critical temperature and pressure of the fluid. Fig. 8 also indicates that the pinch-point temperature difference occurs at closer to the cold end of the evaporator with fluids toluene and cyclohexane, and only small portion of the heat is used for preheating the fluid, when compared to *n*-pentane and especially R245fa.

The results for the ratios of theoretical heat transfer areas compared to net electric power output are shown in Fig. 9(a), (b) and (c). Based on the results approximately 2–4 times larger ratios of heat transfer area to the process power output are needed in the charge air heat utilization compared to the exhaust gas heat utilization, leading to economically more unfeasible processes. The large heat transfer areas in CA utilization can be mainly explained by the smaller temperature differences between the heat source and the working fluid in the evaporator, as well as by the lower cycle efficiency when compared to the EG utilization, due to the lower evaporation temperatures.

It should be noted that the charge air outlet temperatures corresponding to the highest power outputs of ORC, are relatively high compared to typical charge air temperatures after the engine CAC. Thus an additional cooling for charge air would be required after the ORC evaporator. This heat could be used for preheating the ORC working fluid and thus replacing the recuperator or cooled down with the engine cooling loop. Also the pressure losses in charge air should be minimized after the pressure loss has a negative effect on reciprocating engine performance and this should be taken into account when designing the ORC evaporator for charge air utilization.

3.3. The effect of the condensing temperature and the minimum pinch-point temperature difference on the ORC performance

The previously presented results were obtained by using the condensing temperature of 50 °C and the minimum pinch-point temperature difference was set to 15 °C. The effect of minimum pinch-point temperature difference and condensing temperature on process performance in exhaust gas utilization and charge air utilization are presented in Figs. 10(a), (b), 11(a) and 11(b). The selected working fluids are toluene for EG heat recovery and *n*-pentane for CA heat recovery.

The results show that the minimum pinch-point temperature difference and condensing temperature has a significant effect on the process performance. If a low condensing temperature and small temperature differences are used, a greater electric power output is achieved, compared to the higher condensing temperatures and larger temperature differences in the ORC evaporator. Based on the results the relative effect of condensing

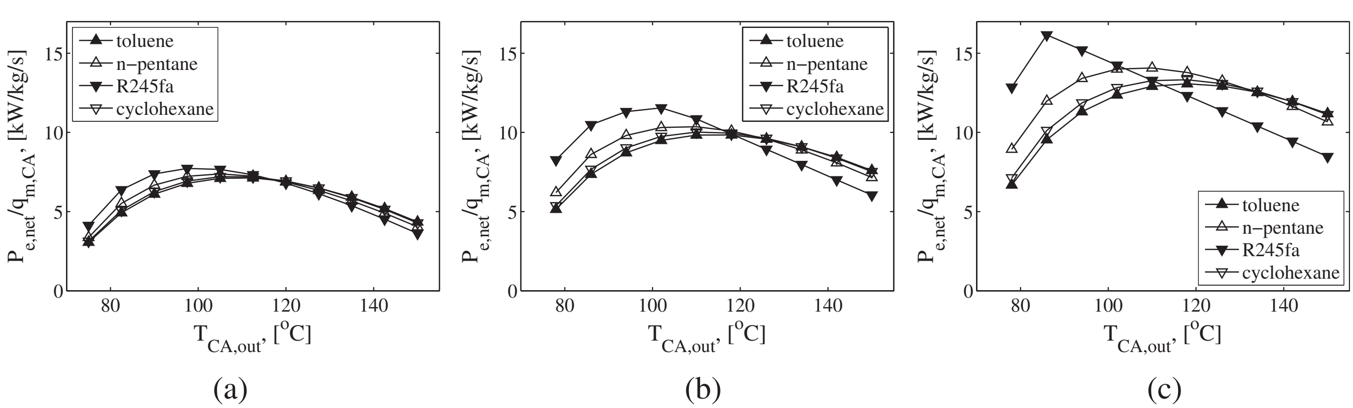

Fig. 7. The effect of charge air temperature at the outlet of the evaporator on the ratio of net power output to charge air mass flow rate for different organic working fluids with charge air inlet temperatures of (a) 180 °C, (b) 200 °C, (c) 220 °C.

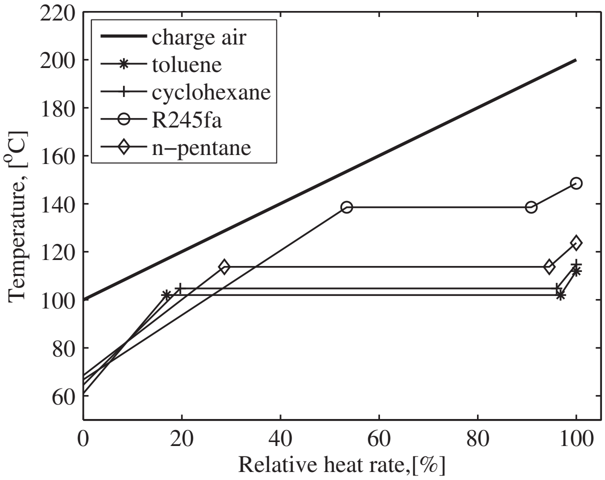

Fig. 8. An example of temperature profiles in the evaporator in 200 °C charge air utilization.

temperature and minimum pinch-point temperature difference on system power output is more significant in the CA heat utilization than in the EG heat utilization, due to the smaller temperature difference between the condenser and the evaporator. Thus, especially the potential to produce electric power from charge air heat is highly related to these above mentioned cycle operating parameters.

4. Case study

A case study of available additional electric power output was performed for engine combined cycles utilizing EG and CA heat. The used computational method is as presented in the previous sections. The minimum pinch-point temperature difference in the ORC evaporator was limited to 15 °C and the condensing temperature of 50 °C was used. The case study was performed for a large industrial gas-fired diesel engine having an electric power output of 16.6 MW. The EG temperature after the turbocharger of the engine is 395 °C and the EG mass flow rate is 27.0 kg/s. The CA temperature after the turbocharger is 210 °C and the CA mass flow rate is 26.2 kg/s. The selected working fluids for the analysis were toluene in EG heat recovery and R245fa in CA heat recovery. The main results are presented in Table 4. The results show that by utilizing the EG heat, a maximum increase of 11.4% in the electric power output was obtained for the selected engine. The CA heat utilization increases the power output of the engine system by 2.4%.

5. Conclusions

The aim was to study the potential for electricity production from the waste heat of large reciprocating engines with the ORC process technology, and hence to increase the efficiency of the engine power plant. A thermodynamic analysis was carried out for working fluids toluene, *n*-pentane, R245fa and cyclohexane and the effect of different process parameters on the cycle performance was studied. The highest potential is related to exhaust gas heat

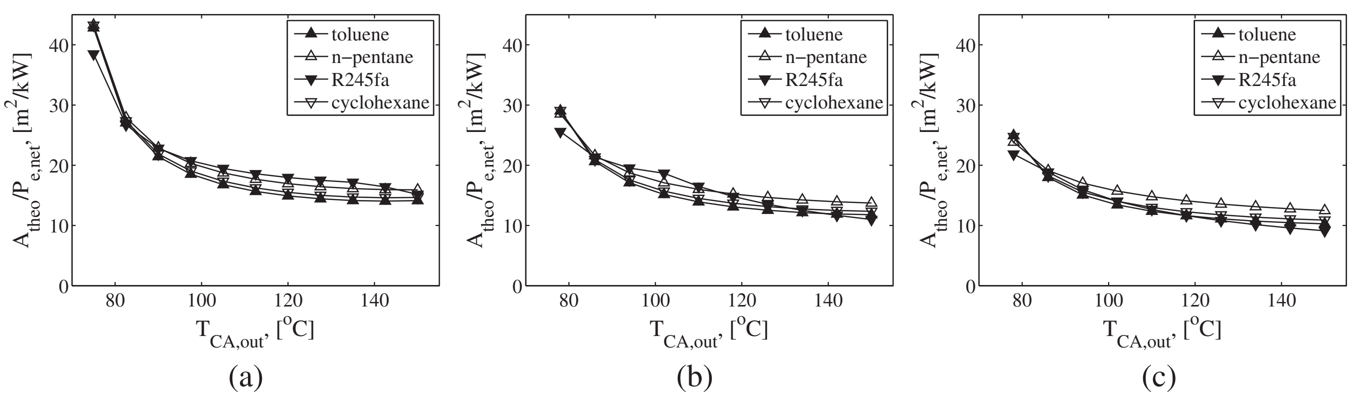

Fig. 9. The effect of charge air temperature at the outlet of the evaporator on the ratio of theoretical heat transfer area to net power output for different organic working fluids with charge air inlet temperatures of (a) 180 °C, (b) 200 °C, (c) 220 °C.

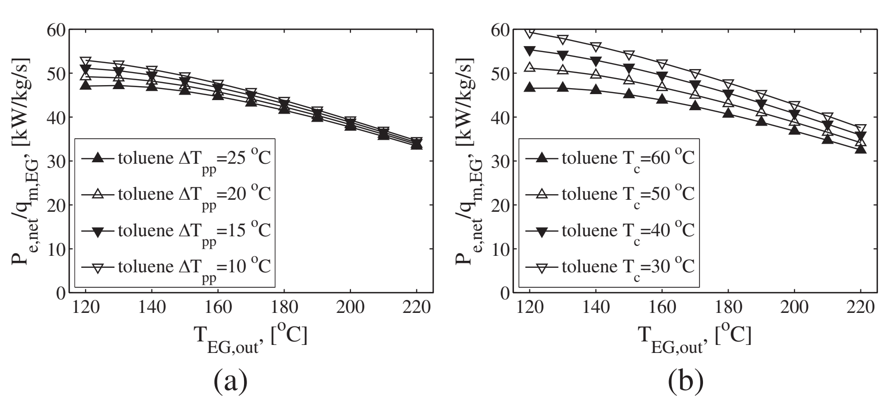

Fig. 10. The effect of pinch-point temperature difference (a), and condensing temperature (b) on ORC power output. The working fluid is toluene and the exhaust gas inlet temperature is 350 °C.

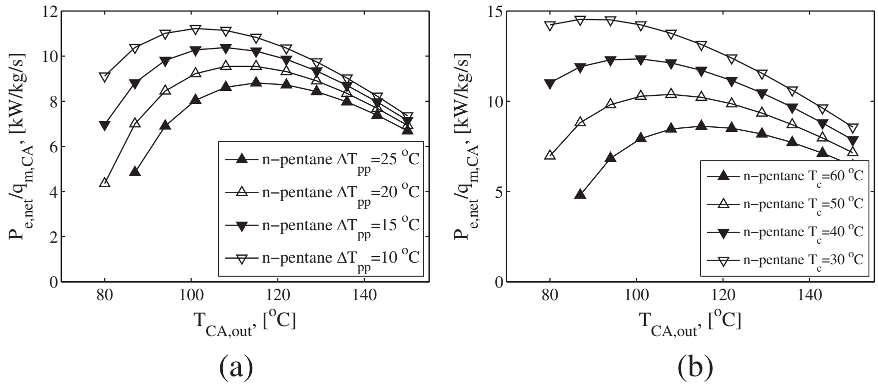

Fig. 11. The effect of pinch-point temperature difference (a), and condensing temperature (b) on ORC power output in charge air heat utilization. The working fluid is *n*-pentane and charge air inlet temperature is 200 °C.

recovery but also charge air heat can be identified as a potential for additional power production. A case study was performed for a 16.6 MW gas-fired diesel engine and the results show that power output of the selected engine can be increased by 11.4% by utilizing exhaust gas heat and 2.4% by utilizing the charge air heat.

Fluids with higher critical temperature, toluene and cyclohexane introduced the highest power outputs in exhaust gas heat recovery. Fluids with lower critical temperature, R245fa and *n*-pentane introduced the highest performances in charge air heat utilization, especially with low charge air outlet temperatures. The ORC process performance is also highly dependent on the heat source temperature, pinch-point temperature difference in the evaporator and the condensing temperature used in the process. The power output increases if a smaller temperature differences are allowed in the heat exchangers, but on the other hand, larger heat transfer areas are needed in the processes. The results show that selection of the working fluid and the process parameters should not be based only on maximizing the power output of the ORC system, but also considering the economic feasibility of the process, by evaluating the ratio of heat transfer area needed and the power output of the process. The results for the ratio between theoretical heat transfer areas and power output show that the exhaust gas utilization would give approximately 2–4 times smaller heat transfer area per produced kW when compared to charge air heat utilization. In charge air heat utilization relatively large heat transfer areas are needed compared to the electric power output achieved, leading to economically more unfeasible processes. A more detailed analysis should be carried out by taking into account a more accurate heat transfer coefficients for each fluid and adopting the effects of using different heat exchanger types.

It should be also highlighted that the pressure losses in charge air and exhaust gas have an effect on the reciprocating engine performance. Thus, a more detailed analysis considering the effect of these pressure losses on reciprocating engine performance as well as studying control strategies and the off-design performance of this type combined cycle power plants are recommended to be carried out in the future. In addition, ORC systems adopting supercritical fluid conditions or having multiple pressure levels could be investigated, in order to better match the working fluid temperature with the heat source, and hence increase the power output of the ORC systems.

Table 4
Case study results.

EG heat recovery 395–130 °C	
Heat rate to evaporator	7921 kW
ORC power output	1895 kW
ORC electric efficiency	23.9%
Increase in power plant electric power output	11.4%
CA heat recovery 210–85 °C	
Heat rate to evaporator	3333 kW
ORC power output	394 kW
ORC electric efficiency	11.8%
Increase in power plant electric power output	2.4%

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